In this research paper, focusing on the basic variables of the gear modeling process and setting its dimensions to see which gears are capable of withstanding transmission operations and its rigidity was done. Because one of the most prominent transmission mechanics is gears, as the types of gears are numerous and common, and one of the most prominent types of gears is the helical gear. The helical gear is one of the most widely used and widespread gears in mechanical fields due to the increase in the contact area during the interlock process, as this increase reduces noise during gear rotation. Three main variables were used to establish the results. The first of which is the pressure angle, the helix deflection angle, and the module number, and they made a number of cases to see which one was able to withstand the movement operations with a proven torque. The results proved that the distortion value in the first case at module 1 was 87×10^{-6} m, while in module 2 the distortion value was 3.75×10^{-6}. The data are useful and important because the values of the stresses that affect the gears must be known by changing the module due to it gives a stronger concept of the extent to which the gears can withstand movement. Pressure angle is one of the basic variables that change the dimensions of wind turbine gears. The value of the greatest stress was 2.13×10^{8} Pa, but at the pressure angle of 20 degrees, the stress value was 1.93×10^{8} Pa. It affects the diameter, stiffness and tensile strength of a wind turbine. The study of this research paper depends on helical gears. It is known that the angles of the helical teeth increase the large contact area between two gears. From the resulting deformation values, it is noted that the deformation value is 4.26×10^{-6} m when the helix angle is 20 degrees

Keyword: Finite element method, helical gear, pressure angle, helix angle, stress analysis

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IDENTIFYING THE INFLUENCE OF DIMENSIONAL PARAMETERS ON THE STRESSES AND DEFORMATIONS OF TWO HELICAL GEARS

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1. Introduction

Since there are many different varieties of gears and they are often used, gears are one of the most important transmission mechanics. The helical gear is one of the most well-known gear types. One of the most common and used gears in mechanical areas is the helical gear. Because of the expansion of the contact surface during the interlocking process, which minimizes noise generated by the rotation of the gears.

Because of the development of simulation engineering for complex physical conditions of mechanical cases and deformations resulting from stresses, it became necessary to know the engineering designs of gears – that were in the form of standards for our time- in the design of large mechanical systems. Where it is necessary to use simulation programs to show and calculate the best dimensions used in mechanical systems that it can be used. There has not been enough research done on such an important topic through which it is possible to find the gap and the lack of correct selection of the parameters of the complex gears.

2. Literature review and problem statement

The paper [1] presented helical pinion wheels transmit mechanical power cogwheels. The results showed that pressure exerted on the tooth often influences various functional limits of a gear. Eight cases were made from *a* to *h*, and the work was on contact motion or movement, and the strength of the existing stresses was not worked out, as the coefficient C0, C1, i.e., the movement settings or the angle of rotation, and its effect on the curvature, and the location of the movement were calculated. The effect of tooth bowing weight on single and herringbone helical material frameworks was investigated by [2] using a restricted component examination. There was a growing need for gears with high load carrying capacity, competency, low cost, low commotion, and low volume. Volume of a helical spline pair is restricted by using RCGA to include profile shift coefficients. The models were for single helical gear and herringbone helical gear. The advancement mechanism uses the imperatives of tooth contact and twisting power as reported by [3]. For low weight and productivity, the full-scale computation of a helical structure pair was simplified. The optimum responses for five combinations of the three objectives were broken down into patterns. The shaft of the helical gear has been improved through volume optimization using coded genetic algorithm and not a simulation. The planar aim for the stuff commotion was chosen by [4] as the top-to-top static transmission blip (PPSTE). A strategy for computing the disappointment likelihood of round and hollow pinion wheels with curved formed (roundabout) teeth along the length was created to twist perseverance. Work was done on the mass and efficiency of the helical gear and the error of the transmission process during movement. The effect of the pressure angle or the gear angle was not addressed only calculations and not simulations also. The arrangement of the issue was acquired by [5] utilizing the numerical contraption of nonparametric measurements, which gives disappointment likelihood computation no matter the intricacy of the anxieties happening at the tooth root during gear activity. Deviated helical cogwheels have been scrutinized for over twenty years. An equipped component is normally stacked (driving/driven side) while the other remaining parts are mostly unloaded (coast side). A new type of gear was worked on, where the failure was studied and the results were extracted in the form of mathematical diagrams, not simulation as well. The parameters of the gear were not studied also. Because of the deviation of the tooth, a nonlinear model was utilized by [6]. Gear boundaries, for example, pressure point, helix point, and so on, influence the conveying limit of stuffed teeth. Here, the research is a simulation without contours details, where the cases of the stiffness of the gear were according to the width of the gear, as we did not address in our work the width because this study has touched upon it. The results showed that the 'exploratory' technique brings twisting pressure closer to the true value as presented by [7]. 10 % ductile pressure and compressive pressure could be reduced by increasing the helical point. This is because the top advantage of time-varying tooth-root pressure switches from HPSTC to LPSTC. In two-fold tooth contact conditions, the increase in equal misalignment might be cut by 18 % throughout the approach phase as shown by [8]. The effects of the stuffed teeth addendum on the improved outcomes for gear full-scale math configuration were evaluated in this study. Where, the research studied the mechanical error and coefficient of friction, hence one tooth or one gear engagement was studied. The results revealed that, by [9], the average score for the advancement while considering just the major plan variables was 2.7085 points, compared to 2.7829 and 2.8830 points, respectively. The impact of improved tooth profile on gear activities is enormous. Forestalling undermining, adjusting of wear and twisting weariness strength, and focus distance change are some of the upsides of profile tooth alterations. Influence of gear tooth addendum and dedendum on the helical gear optimization was studied and not the stresses that it bears as a result of changing the gear model as we did. The impact of profile shift and explicit sliding on plan enhancement of helical gear pairs has also been considered by [10–15] for design optimization with balanced specific sliding and modified tooth profile. In order to determine which gears can endure transmission activities and their stiffness, this study work focuses on the fundamental factors of the gear modeling process and determining its dimensions. There were three key variables utilized. They constructed many examples to determine which one could endure the movement operations with a confirmed torque, starting with the pressure angle, helix deflection angle, and module number. For the design of big mechanical systems, familiarity with the engineering designs of gears – which were in the form of standards for our time- became important as simulation engineering for complicated physical conditions of mechanical cases and deformations resulting from stresses developed. When demonstrating and calculating the optimal size for mechanical systems, simulation software is essential. Too little study has been devoted to this vital subject to allow for the accurate identification of the omission and the determination of the appropriate parameters for the complicated gears. It's true that gears are no longer a hot subject, but in our case, simulation is very important since it reveals numerous flaws and mistakes that would be impossible to detect via experimental effort. This study work is significant because it provides evidence for the parameters with the highest bearing for gear loads and hardness.

3. The aim and objectives of the study

The aim of this study is to identifying the influence of dimensional parameters on the stresses and deformations of two helical gears. This will make it possible to achieve the appropriate selection of helical gears in applications with loads that require a good inspection of the gear parameters to avoid failure as a result of stresses.

To achieve this aim, the following objectives are accomplished:

– to study the effect of module number on the values of stresses and distortions;

– to study the effect of pressure angle on stress and deformation values;

– to study the effect of helix angle on stress and deformation values.

4. Materials and methods of research

4. 1. Object and hypothesis of the study

The importance of simulation at present time is great and it shows to us many defects and errors that cannot be seen with experimental work.

The object of this study is about helical gears which were studied with different dimensions and with a numerical study. In previous research, such variables were not addressed, as the gear was studied in one dimension. In other studies, only the experimental part was studied, and no detailed examination of the numerical part was carried out.

The main hypothesis of the study is to examine helical gears in a range of sizes and from a numerical perspective. As the gear has only ever been investigated in one dimension, such factors have been ignored in earlier study. Whereas the experimental component has been thoroughly investigated in previous works, the numerical component has received less attention.

To study the parameters of helical gears, simulations were carried out under the assumptions that gears have a torque of 8.5 N·m in the counterclockwise direction was applied to the inner part of the first gear in the models to reflect the torque of the motor step. It was connected between the two gears after being fitted from the inside of the other gear. Each scenario makes use of its own set of variables, the first of which is the module number, which ranges from 1 to 2. With regards to the number of teeth, everyone had 30. The normal pressure angles of 20 and 14.5 degrees were utilized, while the inclination angles of the gear or gear in the spiral were 20, 30, and 45 degrees.

The simulation process requires modeling of the body and mesh on the basis of the number of the element, so the case was simplified and the use of a single mesh by two helical gears.

4. 2. Modeling methodology

The model was designed with the Solidworks program, which is a precise engineering design program used in engineering simulation programs, where the models were designed according to the rules of ISO and gears were interlocked using a set of variables to see which one was better in terms of carrying it during use, as shown in Fig. 1.

The benefit of using the Solidworks program lies in designing the gears according to ISO, joining its parameters and exporting it to the ANSYS program as Parasolid file to simulate stress conditions on the geometry that were imported from the Solidworks program. With the use of the Solidworks software design library, the chosen design parameters and in-depth images of the helical gear wheel to be designed were produced. When the cylindrical helical gear is chosen from the toolbox menu of the Solidworks program and the input parameters are chosen, the computer automatically creates the drawing in three-dimensional solid modeling. The Solidworks program's helical gear wheel has been imported into the ANSYS Workbench Finite Elements Package program. It is divided into finite elements using the ANSYS application after meshing.

The simulation using Solidworks software is started after the process of designing the model. It is an engineering program that simulates stress in which a suitable mesh must be made for the purpose of obtaining accurate results that can be compared to practical applications, where the reliability of the mesh and the increase of the mesh until a stable result is reached. The material used is stainless steel. The trihedral mesh was used with a number of elements that reached 410179 and the sizing element was 1 mm. In addition, since the Ansys program is considered one of the trusted programs, the verification is done by means of the mesh independency, as shown in Table 1, where the bone deformation value in this case was stable and was 3.29×10^{-6}, as shown in Fig. 2. Other details are: span angle center is coarse, bounding box diagonal is 88.9060 mm, average surface area is 53.6150 mm^2, and minimum edge length is 0.322150 mm.

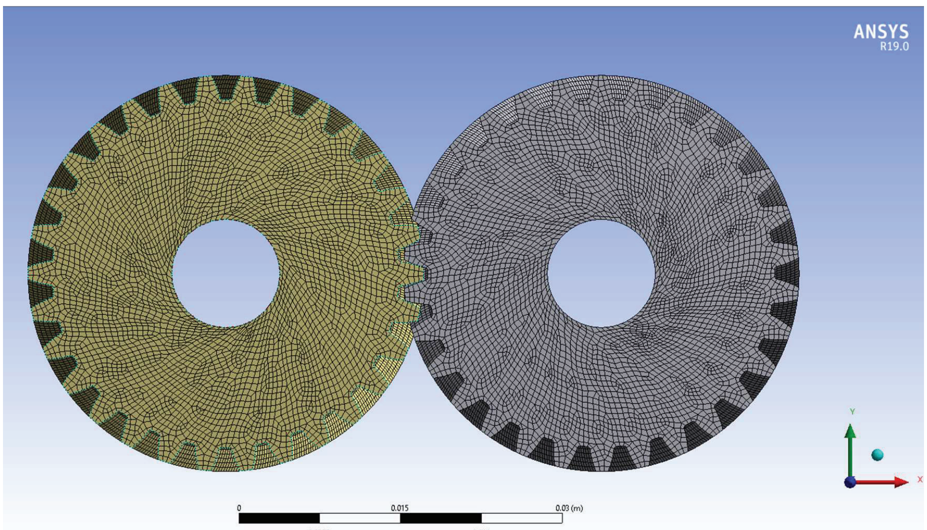


Fig. 2. Geometry mesh

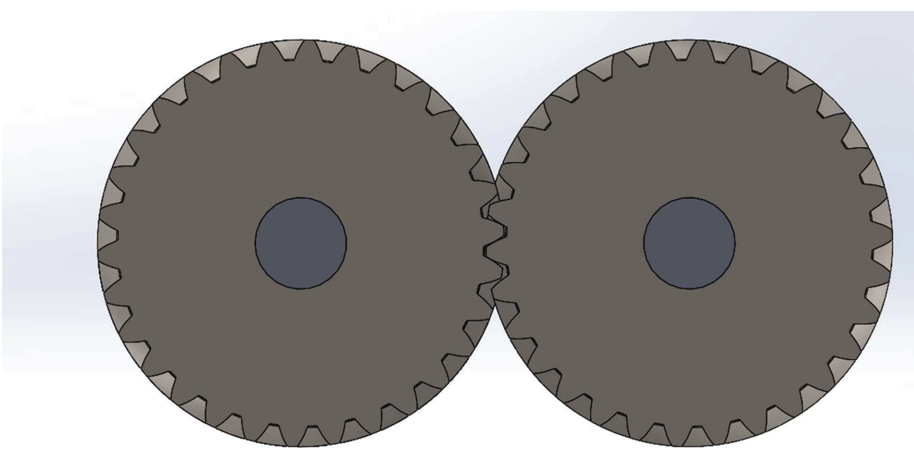


Fig. 1. Geometry design

After the process of knowing the correct type of mesh, the models were simulated, where a torque was shed on the inner part of the first gear, and the amount of this torque was 8.5 N·m in a counterclockwise direction, where this torque represented the torque of the motor step. In the other gear, it was installed from the inside and a connection was placed between the two gears. Different variables were used for each case, the first of which is the module number, where 1, 2 was used. As for the number of teeth, 30 were used in all cases. As for the pressure angle, the standard pressure angles were used, which are 20 degrees and 14.5 degrees, and for the angle of inclination of the gear or gear in the spiral, the angle values were 20, 30, and 45 degrees.

Table 1

Mesh independency

Case	Element	Node	Max. deformation (m)
1	195345	1065379	3.50×10^{-6}
2	223456	1456738	3.35×10^{-6}
3	372645	1652443	3.30×10^{-6}
4	410179	1741223	3.29×10^{-6}

4. 3. Governing equations

The overall equilibrium equations for linear structural static analysis are.

$$[K]\{u\}=\{F\}; \quad (1)$$

or

$$[K]\{u\}=\{F^a\}+\{F^r\}; \quad (2)$$

where $K=\sum_{m=1}^{N}[K_e]$ – total stiffness matrix; $\{u\}$ – nodal displacement vector; N – number of elements; $[K_e]$ – element stiffness matrix (described in Element Library) (may include the element stress stiffness matrix (described in Stress Stiffening)); $\{F^r\}$ – reaction load vector; $\{F^a\}$ – the total applied load vector, is defined by:

$$\{F^a\}=\{F^{nd}\}+\{F^{ac}\}+\sum_{m-1}^{N}\left(\{F_e^{th}\}+\{F_e^{pr}\}\right); \quad (3)$$

where $\{F^{nd}\}$ – applied nodal load vector; $\{F^{ac}\}=-[M]\ \{a_c\}$ acceleration load vector; $[M]=\sum_{m=1}^{N}[M_e]$ – total mass matrix; $[M_e]$ – element mass matrix (described in Derivation of Structural Matrices); $\{a_c\}$ – total acceleration vector (defined in Acceleration Effect); $\{F_e^{th}\}$ – element thermal load vector (described in Derivation of Structural Matrices); $\{F_e^{pr}\}$ – element pressure load vector (described in Derivation of Structural Matrices).

Let's consider a one-element column model that is solely loaded by its own weight to demonstrate the load vectors. Reaction Load Vectors and Applied Load Vectors Although the lower applied gravity load is applied directly to the imposed displacement and so does not create strain, it contributes just as much to the reaction load vector as the higher applied gravity load. In addition, any applied loads on a particular DOF are disregarded if the stiffness for that DOF is 0.

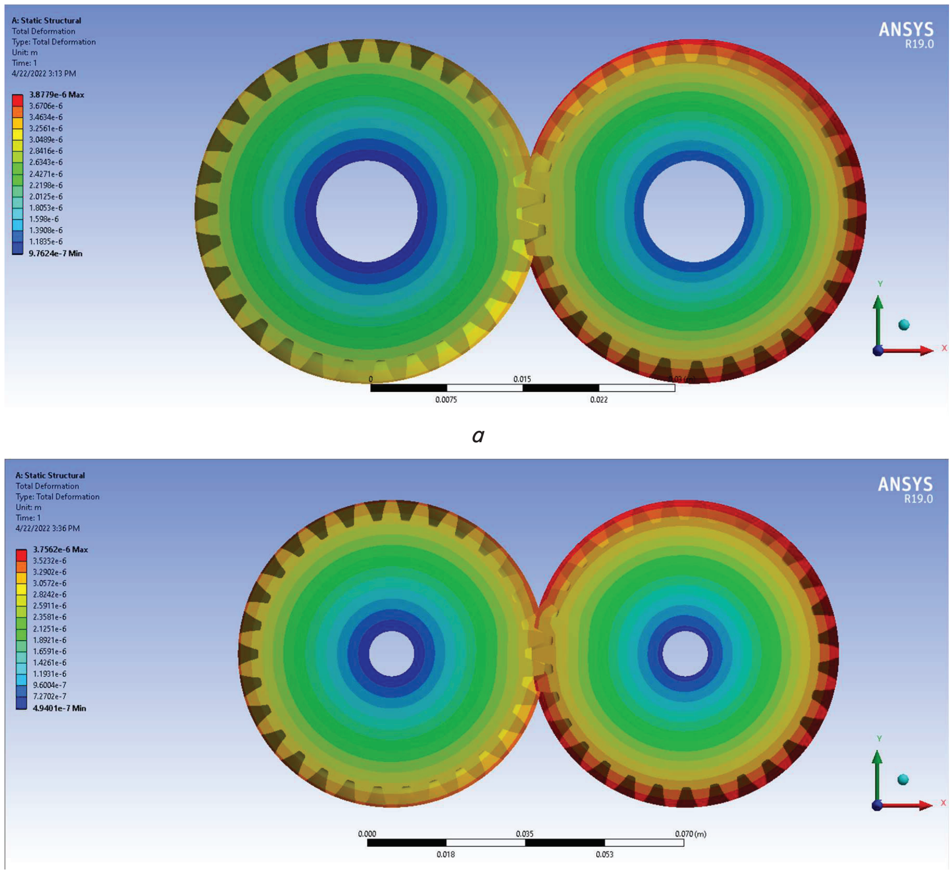
5. Results of static simulation

5. 1. Effect of module number on the values of stresses and distortions

The module number, where 1 and 2 modules were used, is one of the most important dimensional variables that affect the dimensions of the gear. Through the results of distortions, it is noted that the distortion value in the first case at module 1 was 3.87×10^{-6} m. While in module 2, the distortion value was 3.75×10^{-6}, where it is noticed through that module 2 is less distorted compared to the first case, and then it is in terms of its tolerance to distortions as shown in Fig. 3. The following elements were noted during the analysis: due to deformations in the system's elastic domain, it was possible to see the effects of bending teeth in contact, Hertzian contact (local) between the two pairs of teeth in contact, and structural displacements. The stress applied on the part's maximum and minimum total deformation values are depicted in color. Red and blue in this coloring represent the greatest and minimum deformation values, respectively.

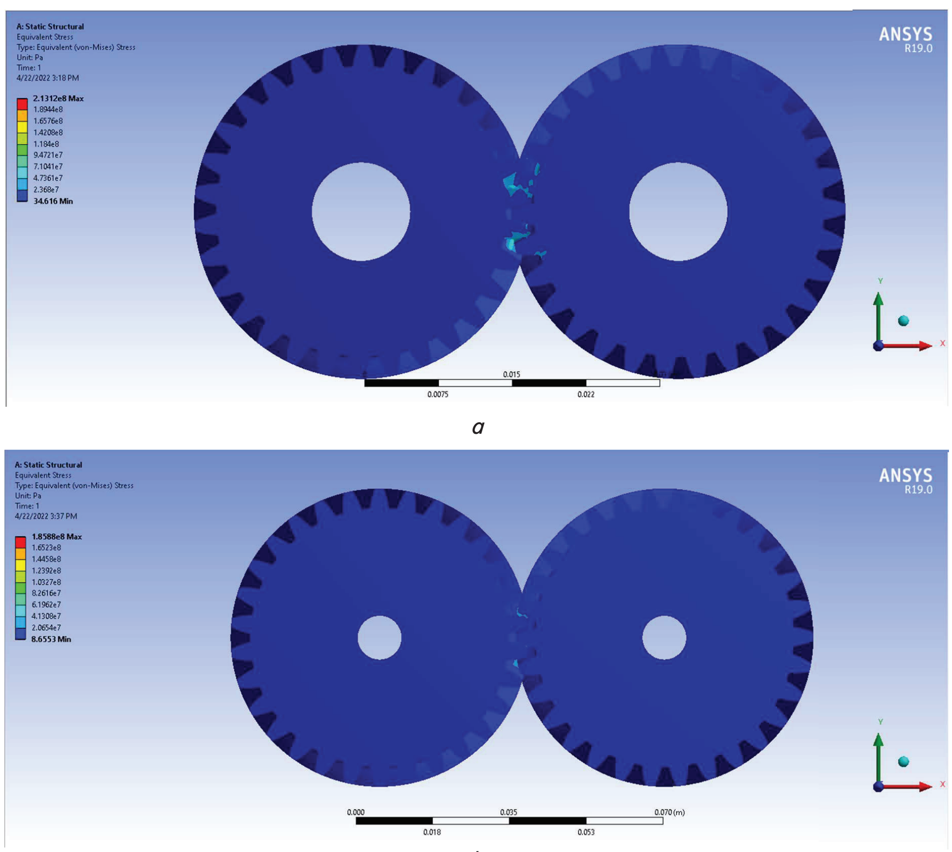
The values of the stresses that affect the gears must be known by changing the module. Because it gives a stronger concept of the extent to which the gears can withstand movement, as shown in Fig. 4.

As shown in Fig. 4, the value of the stresses in module 1 was 2.13×10^{8} Pa, but in module 2, the value of the stresses was 1.85 Pa. The confirmation of module 2 is better than the other case because of the lack of stress on its gears.



a, *b*

Fig. 3. Deformation contour: *a* – module 1; *b* – module 2



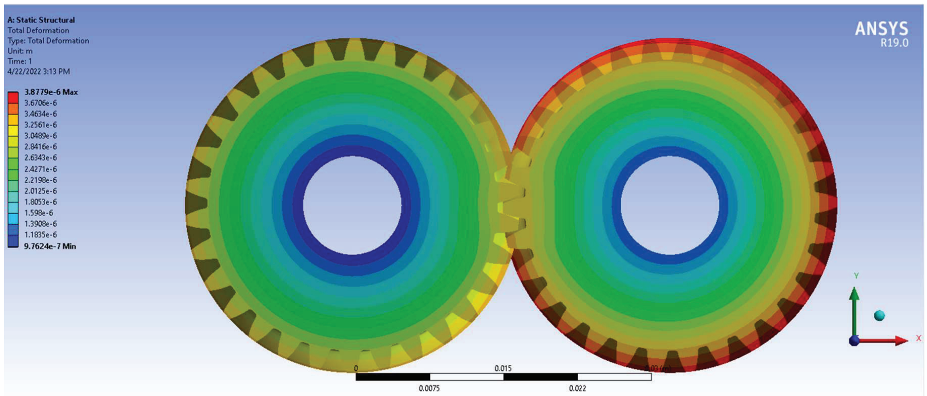
a, *b*

Fig. 4. Stress contour: *a* – module 1; *b* – module 2

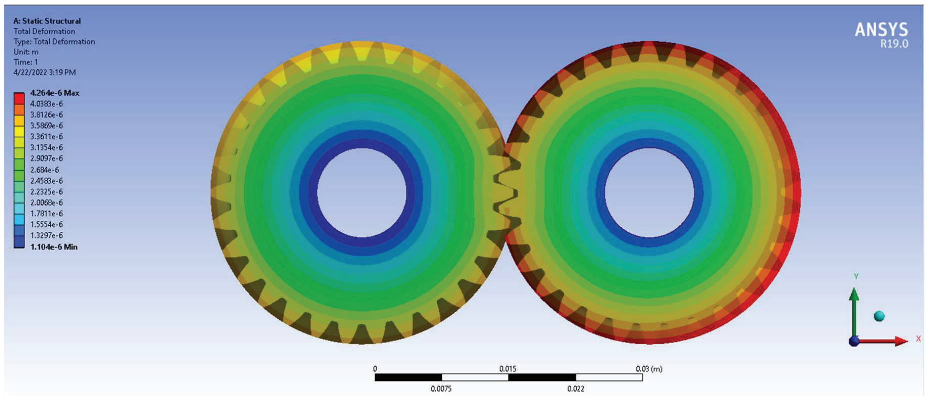
5. 2. Effect of pressure angle on stress and deformation values

The pressure angle is one of the basic variables that change the dimensions of the gears and their engineering structure in terms of their bearing stress and movement. The first type of pressure angle was 14.5 degrees, in which the deformation value was 3.87×10^{-6} m. However, at the other pressure angle, which represents 20 degrees, the deformation value is 4.26×10^{-6} m as shown in Fig. 5, *b*, where it is noted from these Fig. 5, *a*, *b* that the pressure angle 20 is better than the other angle in terms of withstanding the torque. Despite being static, think of it as completing the analysis under actual working settings. To start, let's look at the entire structure's deformation.

The effect of stress at different pressure angles is greatly affected by the difference in the contact area between the first and second gear. Where the stress value in the pressure angle was 14.5 degrees, in which the value of the greatest stress was 2.13×10^{8} Pa, as shown in Fig. 6.

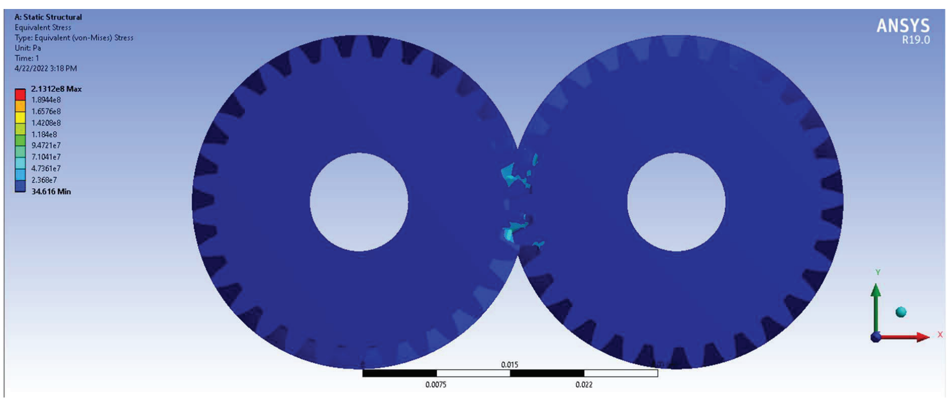


a

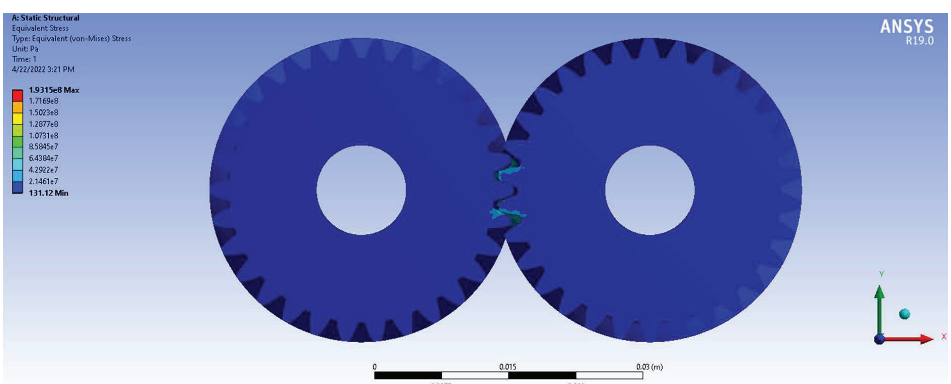


b

Fig. 5. Total deformation: *a* – 14.5 degree ; *b* – 20 degrees



a



b

Fig. 6. Stress contour pressure angle: *a* – 14.5 degrees; *b* – 20 degrees

Nevertheless, at the pressure angle of 20 degrees, the stress value was 1.93×10^8 Pa. It is noted from Fig. 6 that the pressure angle of 20 degrees is better in terms of bearing stress compared to the other angles.

5. 3. Effect of helix angle on stress and deformation values

The study of this research paper depends on helical gears. It is necessary to study the angles of the helical teeth, as it gives a clear idea of the contact area of the meshed gears. It is known that the increase in the angles of the helical teeth increases the large contact area between two gears and reduces the resulting noise. In Fig. 7 of the resulting deformation values, it is noted that the deformation value is 4.26×10^{-6} m when the helix angle is 20 degrees and 3.47×10^{-6} m when the helix angle is 30 degrees, 3.1×10^{-6} m when the helix angle is 30 degrees, and 3.1×10^{-6} m when the helix angle is 45 degrees. Figs show that the increase in the angle of the helix increases the contact area and thus increases the ability of the gear to withstand deformations, as in the angle of the helix of 45 degrees. Along the length of the tooth in contact, a fluctuation in the range of values is seen. Both the contact area and the tooth root exhibit the behavior of the teeth under load.

The value of the stresses is necessary to know the amount of contact between the gears. It is noted from Fig. 8 that the stress value was 1.93×10^8 Pa at the helix angle of 20, 1.86×10^8 Pa at the helix angle of 30, and 1.39×10^8 Pa at the helix angle of 45 degrees.

It is noticed that the increase in the helix angle increases the contact area between the gears and thus reduces the value of the stresses occurring between them. A favorable behavior at the contact pressure with the Hertzian deformation can be seen (small deformation).

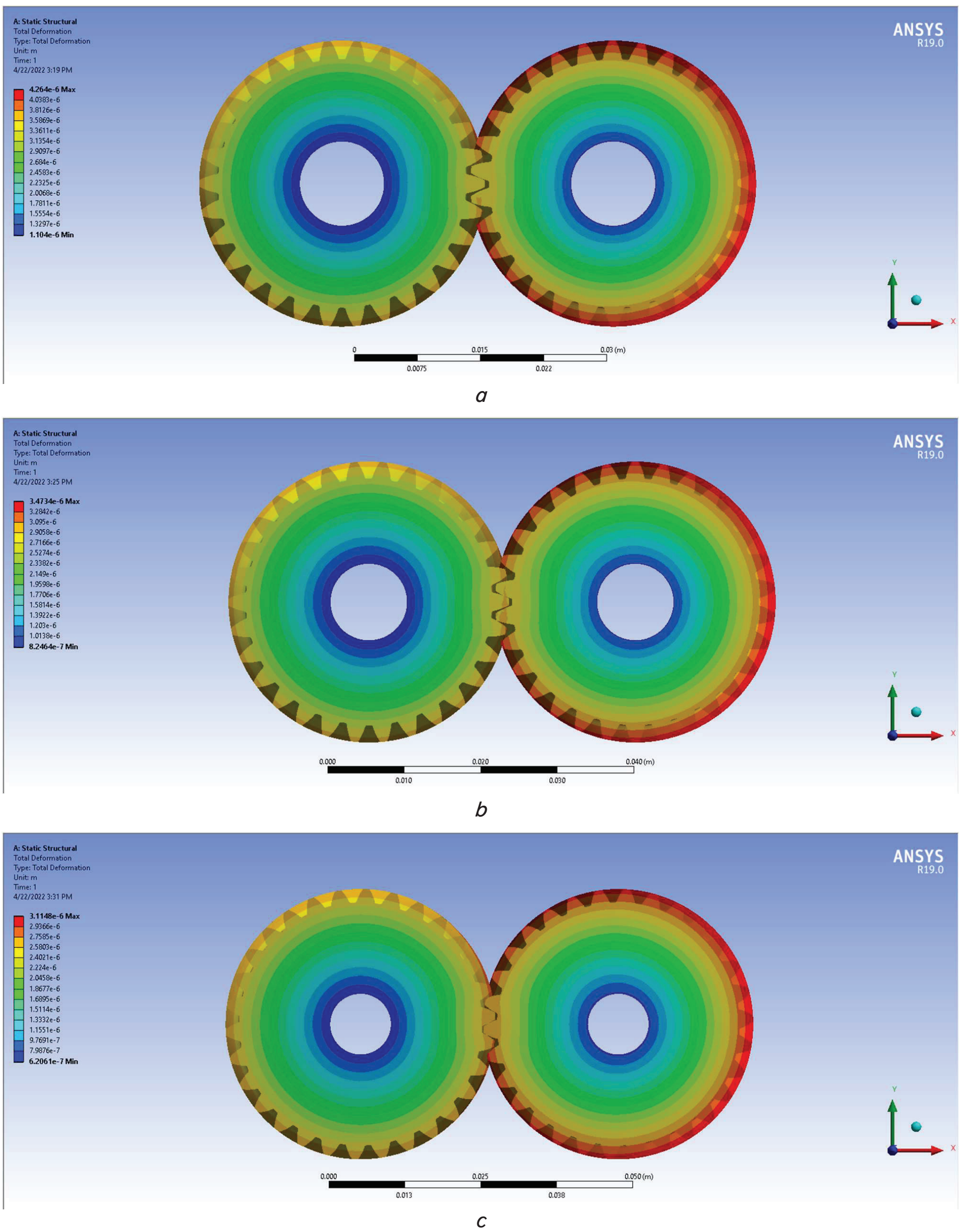
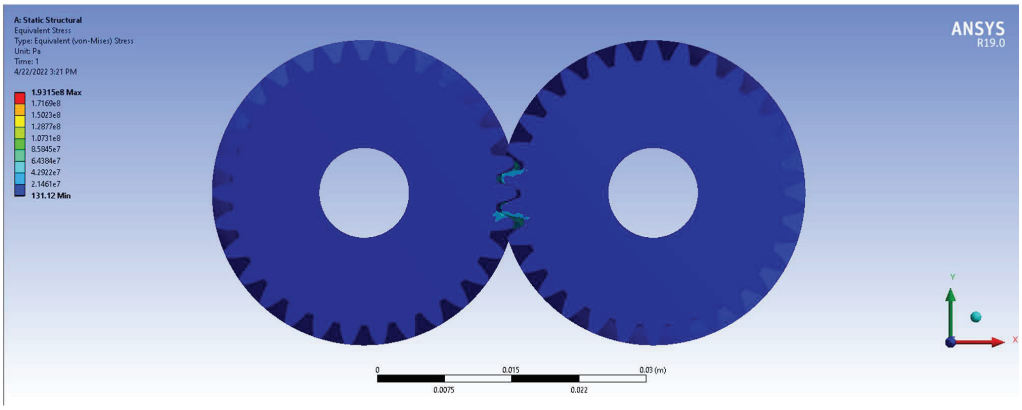
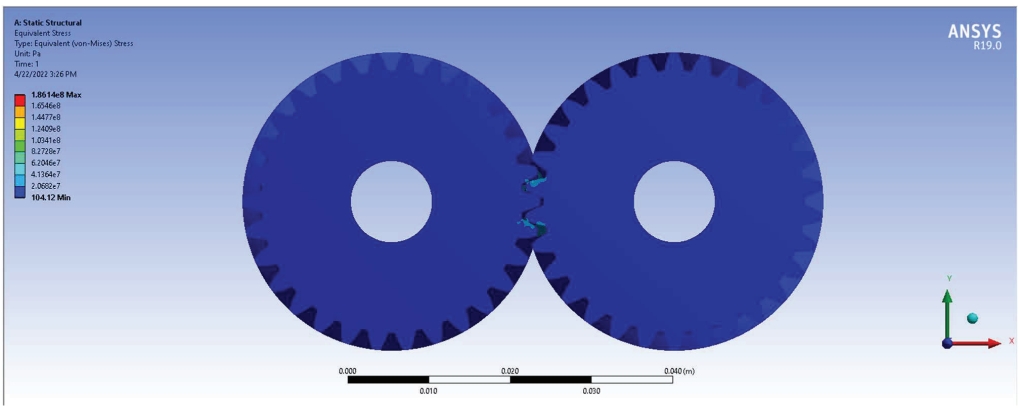


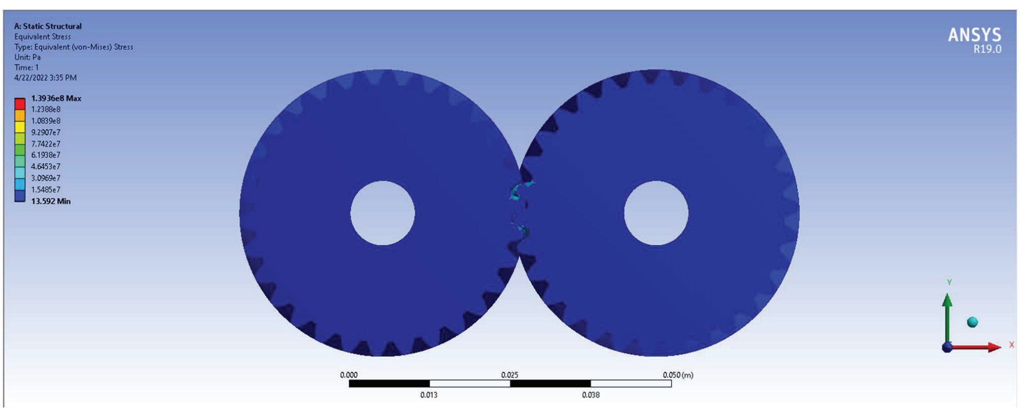
Fig. 7. Deformation contour helix angle: *a* — 20 degree; *b* — 30 degrees; *c* — 14.5 degrees



a



b



c

Fig. 8. Stress contour helix angle: *a* – 20 degrees; *b* – 30 degrees; *c* – 14.5 degrees

6. Discussion of static simulation

In the first scenario, the distortion value at module 1 was 87×10^{-6} m, whereas the distortion value at module 2 was 3.75×10^{-6} m (Fig. 3, 4). Previous results show that the module 2 exhibits less distortion than the second scenario. By altering the module, it is possible to have a better understanding of the stresses that the gears are subject to and how much movement they can tolerate. Due to the absence of pressures on its gears, module 2's confirmation is superior than the other instance.

One of the fundamental factors that affects both the technical design and the gears' size is the pressure angle. In order to calculate the bearing stress and movement of a wind turbine in terms of its diameter, stiffness, and tensile strength, two different kinds of pressure angles were employed. The variation in contact area between first and second gear has a significant impact on the effect of stress at various pressure angles. The maximum stress was 2.13×10^{8} Pa, however with a pressure angle of 20×10^{8} Pa, the maximum stress was 1.93×10^{8} Pa (Fig. 5, 6).

Helical gears are a key component of this research paper's analysis. The enormous contact area between two gears is known to rise as the angles of the helical teeth increase. The deformation value is 4.26×10^{-6} m when the helix angle is 20 degrees, according to the calculated deformation values. Knowing the stresses' values can help to determine how much contact there is between the gears. It should be mentioned that for helix angles of 20 and 30, respectively, the stress value was 1.93×10^{8} Pa and 1.86×10^{8} (Fig. 7, 8).

Because of the lack of such research studies, this encouraged us to do such work, and it was not easy to compare it with other works that were somewhat far from our work, so we just are content with mesh independency.

One of the most important problems encountered is the inability to connect multiple gears to a large extent because the meshing feature is limited in the program.

We do not have supercomputers to solve complex problems, as the multiplicity of gears increases the value of the mesh, and therefore we need supercomputers to solve it.

This study can be developed using a large mesh set to narrate the intrinsic effects in the simulation process.

7. Conclusions

1. The distortion value in the first case at module 1 was 87×10^{-6} m, while in module 2 the distortion value was 3.75×10^{-6} m. The module 2 is less distorted compared to the second case. The values of the stresses that affect the gears must be known by changing the module because it gives a stronger concept of the extent to which the gears can withstand movement. The confirmation of module 2 is better than the other case because of the lack of stresses on its gears.

2. The pressure angle is one of the basic variables that change the dimensions of the gears and their engineering structure. Two types of pressure angles were used to work out the bearing stress and movement of a wind turbine in terms of its diameter, stiffness and tensile strength. The effect of stress at different pressure angles is greatly affected by the difference in the contact area between first and second gear. The value of the greatest stress was 2.13×10^{8} Pa, but at the pressure angle of 20 degrees, the stress value was 1.93×10^{8} Pa.

3. The study of this research paper depends on helical gears. It is known that the increase in the angles of the helical teeth increases the large contact area between two gears. From the resulting deformation values, it is noted that the deformation value is 4.26×10^{-6} m when the helix angle is 20 degrees. The value of the stresses is necessary to know the amount of contact between the gears. It is noted that the stress value was 1.93×10^{8} Pa at the helix angle 20 and 1.86×10^{8} at 30 degrees.

Conflict of interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

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